"Express Mail" mailing label number: EV340866335US

Date of Deposit: March 17, 2004

Our Case No. 10541-1931

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE
APPLICATION FOR UNITED STATES LETTERS PATENT

INVENTOR(S):	Xubin Song
TITLE:	FREQUENCY DOMAIN RIDE CONTROL FOR LOW BANDWIDTH ACTIVE SUSPENSION SYSTEMS
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BACKGROUND

1. Field of the Invention

[0001] The present invention generally relates to a system and method for controlling a vehicle suspension.

2. Description of Related Art

[0002] Generally, people all over the world drive their automobiles to various destinations. In order for these people to enjoy the ride to their destinations the suspensions systems in the automobiles must be stable and as comfortable as possible. Different types of automobiles have various suspension systems, which control the ride and handling performance of the vehicle. For example, some vehicles may have a sport or stiff suspension system that limits movement of its vehicle chassis with respect to the road wheels, but provides less isolation from rough road surfaces. In contrast to the stiff suspension system, some vehicles may have a luxury or soft suspension system that provides a more comfortable ride by isolating the vehicle occupied from the rough road surface, but allowing increased vehicle chassis movement causing a decrease in the handling performance.

[0003] Recently, low-bandwidth active suspension control systems have been developed employing compressible fluid struts and digital displacement pump motors. One key enabling technology of these systems are efficient and effective control algorithms to fully utilize the actuation systems, while avoiding various difficulties of control algorithm implementation. One such difficulty includes developing frequency domain vibration control methods to achieve desired dynamic performance for a specific working frequency range. This frequency range, between zero and up to 30 Hz, provides two significant frequency modes, a body mode

around 1 Hz and a wheel-hop mode around 11 Hz each requiring different suspension control strategies. To implement the control strategies, the control system utilizes the frequency amplitude of the vehicle heave, pitch, and roll to calculate the suspension system adjustment.

[0004] Generally, heave, pitch, and roll frequency information is determined using three body accelerometers. However, it would be advantageous to calculate heave, pitch, and roll frequency information using existing sensors thereby eliminating the need for the three body accelerometers. In view of the above, it is apparent there exists a need for an improved system and method for controlling a suspension system that does not require three body accelerometers.

SUMMARY

[0005] In satisfying the above need, as well as overcoming the enumerated drawbacks and other limitations of the related art, an embodiment of the present invention provides a system for controlling the suspension of a vehicle. The system includes compressible fluid struts as components of vehicle suspension, sensors to measure a strut relative displacement, and a controller configured to determine the frequency amplitude for the heave, pitch, or roll of the vehicle based on the strut relative displacement.

[0006] In another aspect of the present invention, the controller includes a derivative filter to generate a strut relative velocity based on the strut relative displacement. Further, the strut relative velocity is used to calculate a body relative velocity. A first and second frequency amplitude are extracted from the body relative velocity to generate an effective frequency of the suspension. In addition, a desired

strut pressure is calculated based on the effective frequency, the strut relative velocity, and the strut relative displacement. The struts are adjusted in accordance with the desired strut pressure to improve vehicle suspension performance.

[0007] Further objects, features and advantages of this invention will become readily apparent to persons skilled in the art after a review of the following description, with reference to the drawings and claims that are appended to and form a part of this specification.

BRIEF DESCRIPTION OF THE DRAWINGS

[0008] Figure 1 is a schematic view of a vehicle having a system for controlling a suspension system in accordance with the present invention;

[0009] Figure 2 is a block diagram of an algorithm for controlling a suspension system in accordance with the present invention;

[0010] Figure 3 is a block diagram of an algorithm for deriving three body relative velocities from four strut relative displacements in accordance with the present invention;

[0011] Figure 4 is a block diagram of a frequency decoding algorithm in accordance with the present invention;

[0012] Figure 5 is a block diagram of an algorithm for extracting frequency amplitude in a frequency decoding algorithm in accordance with the present invention;

[0013] Figure 6 is a plot of a sample control strategy for heave stiffness control; and

[0014] Figure 7 is a block diagram of an algorithm for determining desired strut pressure in accordance with the present invention.

DETAILED DESCRIPTION

[0015] Referring now to Figure 1, a system 12 for controlling the suspension of the vehicle 10 and embodying the principles of the present invention is provided. The system 12 includes an electronic control unit 16, a digital displacement pump motor (DDPM) 18, compressible fluid struts (CFS) 14, and displacement sensors 15. A suspension system of this general type is generally disclosed in U.S. Patent Application No. 10/688,095, filed on October 17, 2003, which is hereby incorporated by reference.

[0016] Electronic control unit 16 interfaces with the displacement sensors 15 to collect strut relative displacement information. The strut displacement sensors are of the type well known in the industry and therefore need not be discussed in greater detail herein. Utilizing the strut relative displacement information, the electronic control unit 16 selects a control strategy to optimize the suspension performance and calculates the desired strut pressure information to implement the control strategy. The desired strut pressure is utilized to operate the DDPM 18 thereby tuning the stiffness and damping characteristics of each compressible fluid strut 14 in accordance with the control strategy.

[0017] Now referring to Figure 2, the displacement sensors 15 provide the strut relative displacement 22 to a control algorithm 20 contained in the electronic control unit 16. Block 24 receives the strut relative displacement signals and converts the strut relative displacement signals to body relative velocities 26. In addition, block 24 also generates strut relative velocities 25 to be used in calculating the desired strut pressure 34, 36, 38. Block 28 receives the body relative velocities 26 and performs a frequency decoding algorithm to generate the effective

frequencies 30. Block 32 then generates the desired strut pressure 34, 36, 38 based on the strut relative displacement 22, the strut relative velocity 25, and the effective frequencies 30. The desired strut pressure 34, 36, 38 is received by block 40 to calculate the combined desired strut pressure 42 for each strut 14. The combined desired strut pressure 42 is provided to the digital displacement pump motor 18 to effectuate a desired control strategy by adjusting the pressure in each strut 14. Various portions of the control algorithm 20 will be discussed in more detail below.

[0018] Now referring to Figure 3, the details of block 24 are provided. The strut relative displacement signals 22 (D_{lf}, D_{lr}, D_{rf} and D_{rr}) are received by the derivative filter 50, and the derivative filter 50 generates the strut relative velocities 25 (Vs_{lf}, Vs_{lr}, Vs_{rf}, and Vs_{rr}). The strut relative velocities 25 are independently used to calculate the desired strut pressure as discussed later. Further, the strut relative velocities 25 are received by block 53 to generate the body relative velocities 26, or more specifically the body relative heave, pitch, and roll velocity (V_h, V_p and V_r). For a specific vehicle, wheelbase (L) and tread (t) are known and used to calculate the body relative heave, pitch and roll velocities according to the relationship $V_h=(V_{lf}+V_{lr}+V_{rf}+V_{rr})/4$, $V_p=(V_{lf}-V_{lr}+V_{rf}-V_{rr})/(2*L)$, and $V_r=(V_{lf}+V_{lr}-V_{rf}-V_{rr})/(2*t)$.

[0019] After the body relative velocity V_i (i=h, p and r) is calculated, each signal can be used to extract the effective frequency ω_{ie1}(i=h,p,r) for ride control. Now referring to Figure 4, the frequency decoding algorithm 28 is applied at the vehicle body mode frequency range. Accordingly, the body relative velocity 26 is provided to a high-pass filter 60 and a low-pass filter 62. The vehicle body mode frequency is ω_1 ($=2\pi f_1$), therefore, a lower frequency ω_0 (about two or three times

less than ω_1) can be selected, along with an intermediate frequency ω_{01} between ω_0 and ω_1. These frequencies can be used as break frequencies for the high-pass filter 60 and the low-pass filter 62. The high-pass filtered body relative velocity 61 is used to extract a first frequency amplitude 65 (A_1) at the selected frequency ω_1, as denoted by block 64. Similarly, the low-pass filtered body relative velocity 63 is used to extract a second frequency amplitude 67 (A_0) at the selected frequency ω_0, as denoted by block 66. In block 68, the first frequency amplitude 65 in the second frequency amplitude 67 are combined according to the relationship A_1/A_0 to generate the effective frequency 30.

[0020] Now referring to Figure 5, a description of the algorithm to extract the frequency amplitude at the selected frequency such as in blocks 64 and 66, is provided in reference to selection of the first frequency amplitude 65 (A_1). The high-pass filtered body relative velocity 61 is provided to a washout filter in block 70. The washout filter modifies the high-pass filtered body relative velocity 61 according to certain washout factors 76. The selected frequency 80 (ω_1), along with the result of the washout filter 70, is provided to a band-pass filter in block 72. The results from the band-pass filter 72 and the washout filter 70 are provided to an integrator 74. The result of the integrator 74 is provided, along with the result of the band-pass filter 72 and the selected frequency 80, to a modal generator in block 78. Utilizing the selected frequency information 80 the modal generator result is provided to a smoothing filter 82, which results in the frequency amplitude 65 (A_1).

[0021] Similarly, the above-described algorithm to extract the frequency amplitude at a selected frequency may be applied to the second frequency amplitude 67 (A_0) in the same manner.

[0022] Referring again to Figure 4, the effective frequency ω_{ie1}(i=h,p,r) is used for integrating different control strategies required for different frequency ranges. Similarly, the above procedure can be applied to the frequency range around the wheel-hop mode frequency ω_{ie2}(i=h,p,r). For illustrative purposes the control algorithm for the low-band-width active suspension system is provided.

[0023] For the low bandwidth active suspension, a bandwidth of 5 to 7Hz is targeted due to the limited capability of the DDPM with a limited power supply. Therefore, if the suspension dynamics dominate in the frequency range beyond the bandwidth, the control algorithm will set the DDPM to idle to save power and let the CFS work in a passive state. If the effective frequencies of the suspension dynamics are less than the bandwidth, the control algorithm can select different strategies to better isolate the vehicle body from the subjected vibrations. Those strategies can be stiff stiffness, soft stiffness, soft rebound damping, hard compression damping or variations thereon. In addition, a traditional passive shock absorber damping capability exists in the CFS, such as, hard damping for rebound and soft damping for compression.

[0024] Based on the effective frequencies ω_{ie1} and ω_{ie2} (i=h,p,r), strategy mappings can be determined for stiffness control and damping tuning with different effective frequencies as described in Table1 below. For example, if the heave body mode is 1.4Hz, then the ω_{he1}-based strategy mapping can be –1 (representing stiff stiffness) for ω_{he1} less than 0.9Hz, 1 for ω_{he1} near 1.4Hz (and beyond), and a linearly interpolated value (or other curves) for ω_{he1} between 0.9 and 1.4Hz. The control signals may be reduced beyond the given bandwidth by: (1) Directly forcing the ω_{he1}-based strategy mapping to close to 0 if ω_{he1} is close to 5 to 7Hz and 0 beyond the

bandwidth, (2) Using ω_{he2} to identify the high frequencies so that the ω_{he2}-based strategy mapping is 1 below 5 to 7Hz and becomes 0 beyond the bandwidth. The product of two strategy mappings, ω_{he1} 84 and ω_{he2} 86, for the stiffness control are shown in Figure 6. Similarly the strategy mappings for heave damping can be properly derived from Table 1.

Table 1

	Effective Freq Range	**Adopted Control Strategy**
Ride Control (i.e., Ride Comfort)	Low	Stiff Stiffness and Hard Compression Damping
	Body Mode	Small Stiffness
	< Bandwidth	Small Stiffness and Soft Damping
	> Bandwidth	Passive Suspension (i.e., idle DDPM and no valve control)

[0025] Now referring to Figure 7, the desired strut pressure algorithm 32 is provided in more detail. The strut relative displacements 22 are provided to the transfer function $f(D_i)$ as provided in block 88. Further, $f(D_i)$ (i=lf, lr, rf and rr) is a function of the strut relative displacements, always no less than zero, and the outputs are desired pressures for each of the CFS. The strategy mapping is also used to decide whether a stiff or soft stiffness should be required for the feedback.

[0026] The effective frequency 30 (ω_{ie1} and ω_{ie2}) is provided to the strategy mapping for stiffness heave control as denoted by block 90. In block 92, the product of the transfer function from block 88 and the strategy mapping for stiffness heave control from block 90 is used to generate the desired strut stiffness heave pressure 93. The strut relative velocity 22 is provided to the transfer function $f(V_h)$ as provided in block 106. Effective frequency 30 (ω_{ie1} and ω_{ie2}) is provided to the strategy

mapping for heave damping control as denoted by block 108. In block 110, the product of the transfer function from block 106 and the strategy mapping for heave damping control from block 108 is used to generate the desired strut heave damping pressure 111. The desired strut stiffness heave pressure 93 and the desired strut heave damping pressure 111 are combined in block 112 to generate the desired strut heave pressure 34.

[0027] For pitch control, the strut pitch relative velocity from the strut relative velocity 22 is provided to the transfer function $f(V_p, L/2)$, where L is the wheelbase, as provided in block 94. The effective frequency 30 (ω_{he1} and ω_{he2}) is provided to the strategy mapping for pitch control as denoted by block 96. In block 98, the product of the transfer function from block 94 and the strategy mapping for pitch control from block 96 is used to generate the desired strut pitch pressure 36.

[0028] Similarly, for roll control, the strut roll relative velocity from the strut relative velocity 22 is provided to the transfer function $f(V_p, t/2)$, where t is the tread, as provided in block 100. The effective frequency 30 (ω_{he1} and ω_{he2}) is provided to the strategy mapping for roll control as denoted by block 102. In block 104, the product of the transfer function from block 100 and the strategy mapping for roll control from block 102 is used to generate the desired strut roll pressure 38.

[0029] As a person skilled in the art will readily appreciate, the above description is meant as an illustration of implementation of the principles this invention. This description is not intended to limit the scope or application of this invention in that the invention is susceptible to modification, variation and change, without departing from spirit of this invention, as defined in the following claims.